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THERMODYNAMIC ANALYSIS OF A TIN-STEAMED LMMHD SYSTEM.(U)
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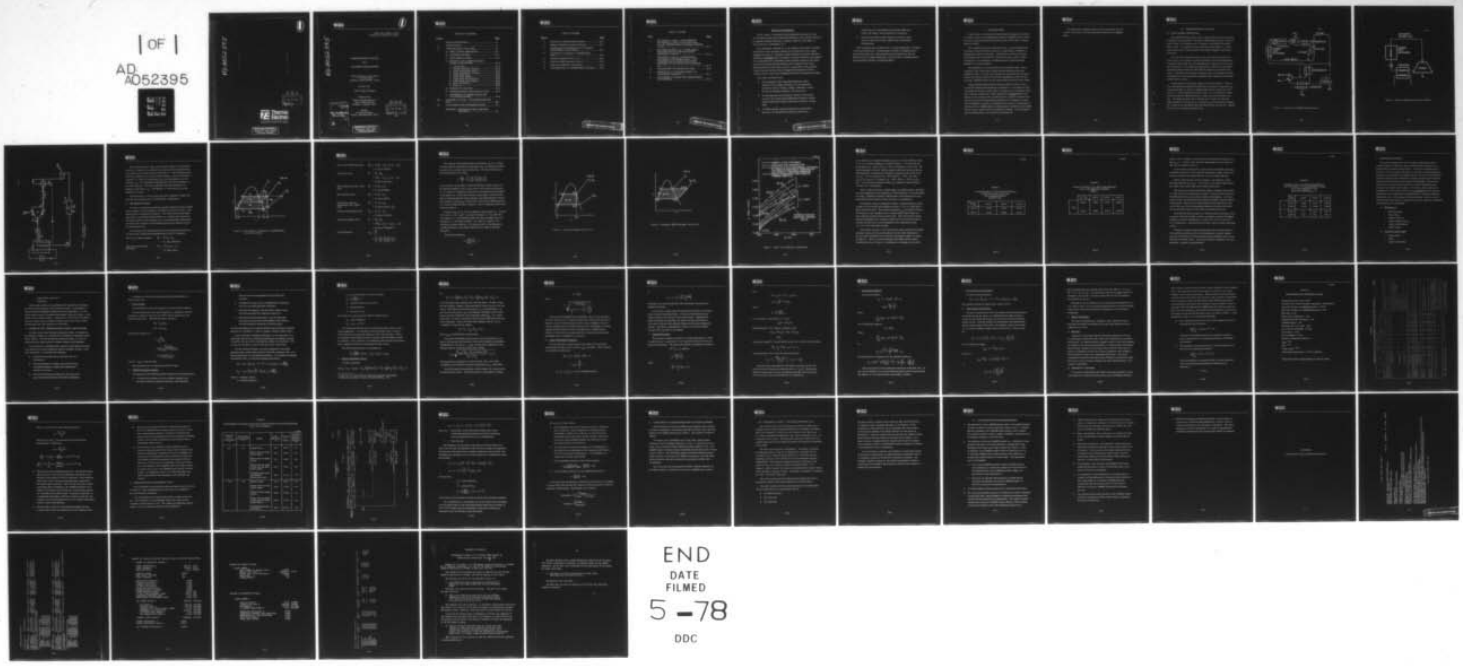

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THERMODYNAMIC ANALYSIS
OF A
TIN-STEAM LMMHD SYSTEM

Thermo Electron Corporation
101 First Avenue
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29 July 1977

Final Technical Report

Prepared For

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TABLE OF CONTENTS

LIST OF FIGURES

Thermo Electron Corporation

LIST OF TABLES

EXECUTIVE SUMMARY

In this report, we evaluate the thermodynamic analyses and data of a tin-steam liquid metal magnetohydrodynamic power system proposed by Argonne National Laboratory, Argonne, Illinois, to the Office of Naval Research, Washington, D. C.

Our evaluations consisted of: (1) an analysis of the ideal or limiting performance of the proposed system; (2) a comparison of the results obtained by means of a computer model developed by Argonne with those obtained by simplified hand calculations by Thermo Electron Corporation; (3) an analysis of the sensitivities of the computer results to the values of key but uncertain input parameters; (4) an analysis of the major efficiency losses in the system; and (5) a comparison of the proposed system with one that does not use magnetohydrodynamics but does require effective steam heat management (reheat and feedwater heating).

Our major conclusions are:

- The advantages of the magnetohydrodynamic steam system must be judged primarily from the standpoint of features such as volume, weight, reliability, amount and size of rotating machinery, and noise level.
- An important goal of the program should be the development of a pure liquid metal magnetohydrodynamic system (without bottoming turbine) with steam or another working fluid.
- A 1-MW prototype system should not be constructed at this time, but consideration should be given to the

construction of a test facility to develop the MHD generator, the mixer, and the separator components.

- The program should include analyses of specific applications, which can provide a guide to the technological developments.

The evaluation team consisted of Dr. George Hatsopoulos, President of Thermo Electron; Dr. Joseph Kestin, Professor of Engineering at Brown University; Drs. Dean Morgan and Fred Huffman of Thermo Electron; and Dr. Elias Gyftopoulos, Ford Professor of Engineering at Massachusetts Institute of Technology (MIT).

I. INTRODUCTION

In this report, we evaluate the thermodynamic analyses and data of a tin-steam liquid metal magnetohydrodynamic (LMMHD) power system proposed by Argonne National Laboratory (ANL) to the Office of Naval Research (ONR).

The evaluation effort was supervised by Dr. George Hatsopoulos, President of Thermo Electron, and Dr. Joseph Kestin, Professor of Engineering at Brown University. Analyses were performed by Drs. Dean Morgan and Fred Huffman of Thermo Electron, and Dr. Elias Gyftopoulos, Ford Professor of Engineering at the Massachusetts Institute of Technology (MIT).

The guidelines for the evaluation were provided by ONR. Some members of the review panel visited the ANL facilities devoted to LMMHD on April 14, 1977, and discussed at length with Drs. Michael Petrick and Edward Pierson of ANL the computational model for the tin-steam LMMHD system and the experimental data used in the model.

Using published information provided by ONR and ANL, we performed the following thermodynamic evaluations: (1) an analysis of the reversible performance of the proposed system; (2) a comparison of the ANL computer results with results obtained by <u>simplified</u> hand calculations; (3) an analysis of the sensitivity of the computer results to the values of key but uncertain input parameters; (4) an analysis of major irreversibilities by means of availability flows; and (5) a comparison of efficiency of the LMMHD-Steam Turbine System with that of an all steam system with or without reheat and feedwater heating. Our conclusions and recommendations are summarized in Section IV.

Throughout the evaluation effort we have enjoyed the full and cordial cooperation of all ANL personnel involved in the LMMHD effort.

II. THERMODYNAMIC ANALYSES

A. BRIEF SYSTEM DESCRIPTION

The LMMHD Rankine System under consideration has been described in detail in a report entitled "Tin-Steam MHD Power System," Volume I, by M. Petrick, R. Hantman, and T. Kassner, ANL/ENG-76-02, September 1975. The system is shown schematically in Figure 1. It has been proposed as an improvement of the standard Rankine cycle shown in Figure 2.

From the thermodynamic point of view, the basic idea of the proposed system is to modify the standard Rankine cycle by allowing steam to expand isothermally from the saturation line outwards in order to improve efficiency. The isothermal expansion is carried out in the MHD generator against molten tin with which the steam has previously been mixed. The steam transmits the expansion work to the molten tin which in turn performs work against the magnetic field created by an external, useful electric current.

A line diagram of the power system is shown in Figure 3. We see from this figure that steam and molten tin are heated to temperature T_h in heat exchanger, h, which is supplied with primary steam from heat source, P. After passing through the mixer, m, the tin-steam mixture passes through the MHD generator, g, which performs external work, W_m. The MHD generator can be thought of as a device in which the steam expands from pressure p_1 to pressure p_2 performing work against the molten tin and simultaneously absorbing heat Q_2 from the tin. Ideally, this occurs at constant temperature T_h.

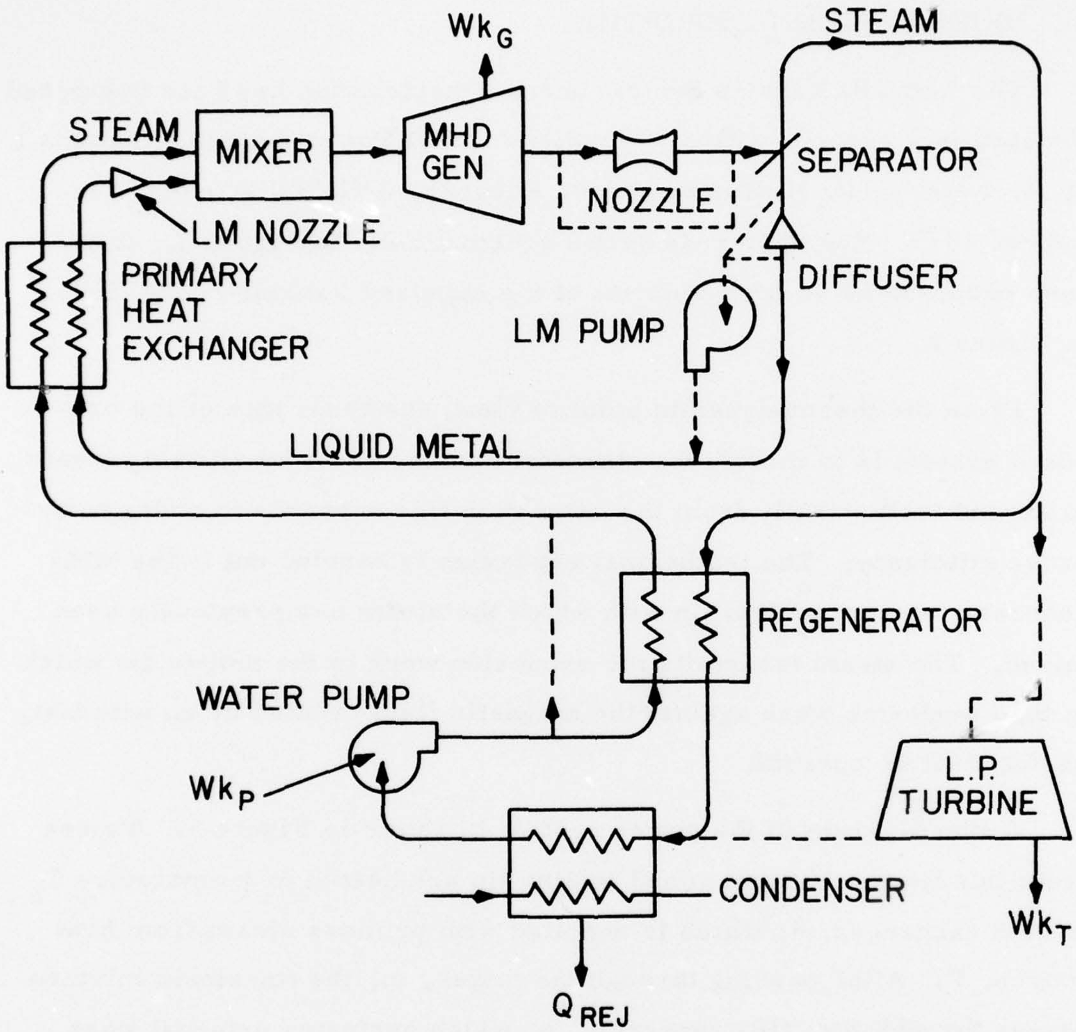

Figure 1. Schematic of LMMHD Rankine System

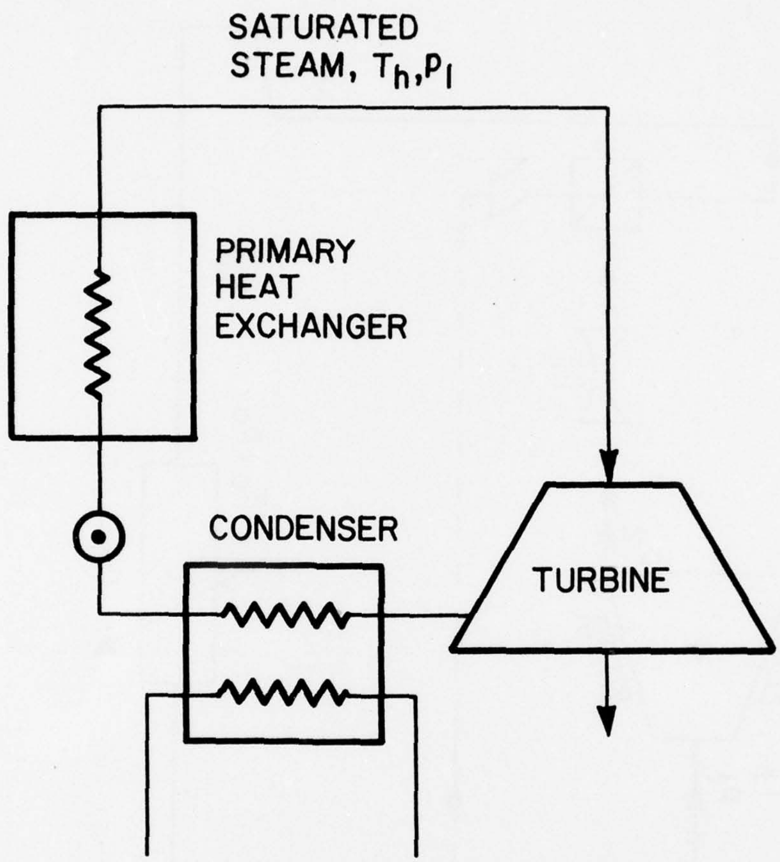

Figure 2. Standard Rankine-Cycle Power System

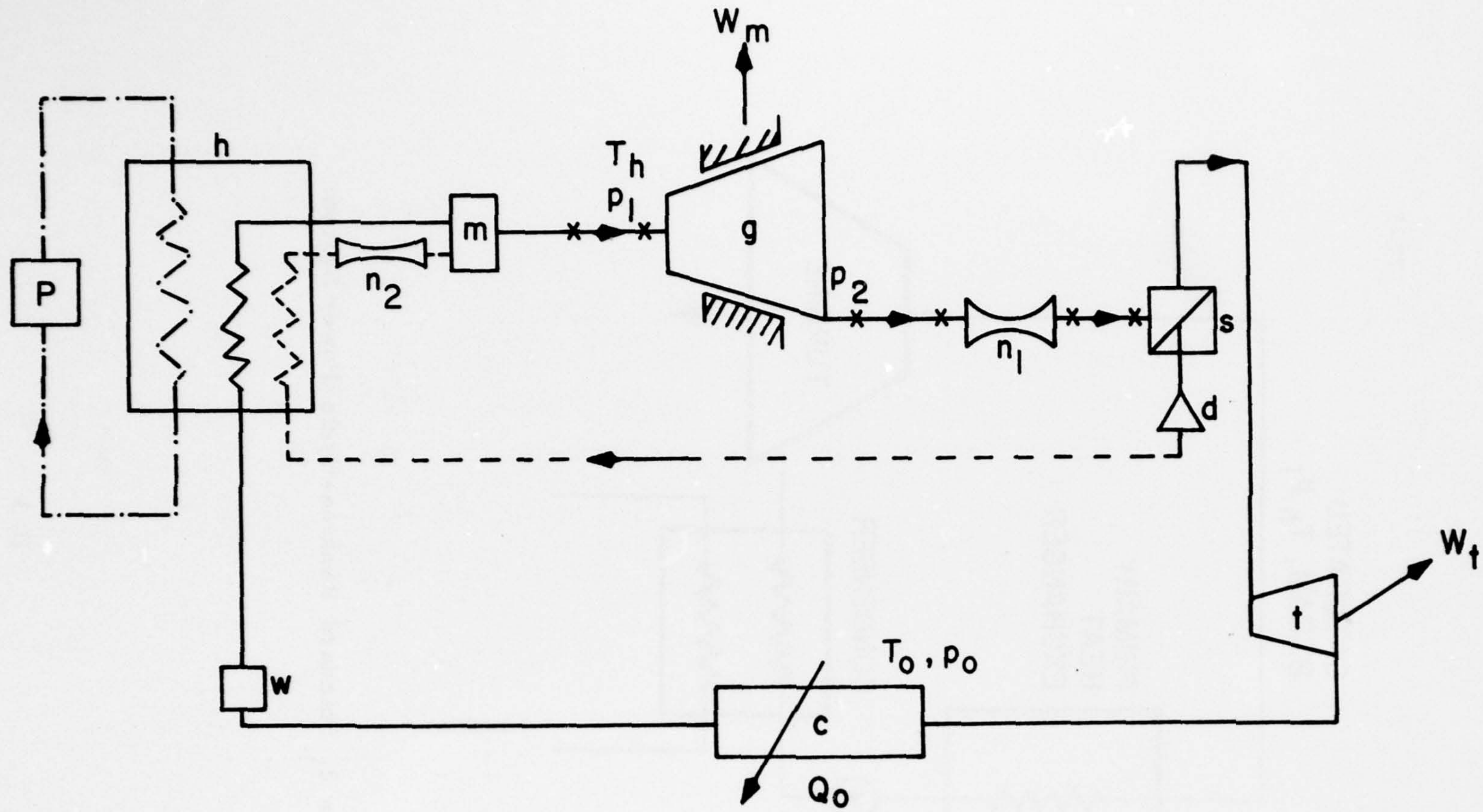

Figure 3. Line Diagram of Simplified Form of Proposed LMMHD-Steam Power System

After performing work W_m, the tin-steam mixture is accelerated in nozzle n_1 (which can be replaced by a pump) and then separated into a tin stream and a steam stream in separator, s. The separated steam performs additional work, W_t, in steam turbine, t, and is condensed in condenser, c, where it rejects heat, Q_o, at a temperature, T_o, and pressure, p_o. The condensate is pumped back into heat exchanger, h, via water pump, w. The tin is returned to the heat exchanger via diffuser, d, and nozzle, n_2. The diffuser, d, and nozzle, n_2, can be replaced by a pump.

In what follows, a work or heat rate will be denoted by using a dot over the corresponding work or heat symbol, respectively.

B. REVERSIBLE MODEL

In order to evaluate the maximum possible efficiency of the proposed power system, we will consider a reversible cycle for the steam. It is shown in the T-s diagram of Figure 4. The work process in the MHD generator is represented by the isotherm 12, and the work process in the steam turbine by the isentropic 24. It is assumed that regeneration is used for the maximum cycle efficiency; this occurs between T_4 and T_5 on the steam side.

The various work and heat rates can now be expressed in terms of the steam flow rate, $\dot{m}$, and appropriate areas of the T-s diagram:

Work rate of steam turbine:
$$\dot{W}_t = \dot{m}\,(h_2 - h_4) = \dot{m}\,(\text{area } 2456792);$$

Heat rate added to MHD generator:
$$\dot{Q}_2 = \dot{m}\,T_h\,(s_2 - s_1) = \dot{m}\,(\text{area } 12bc1);$$

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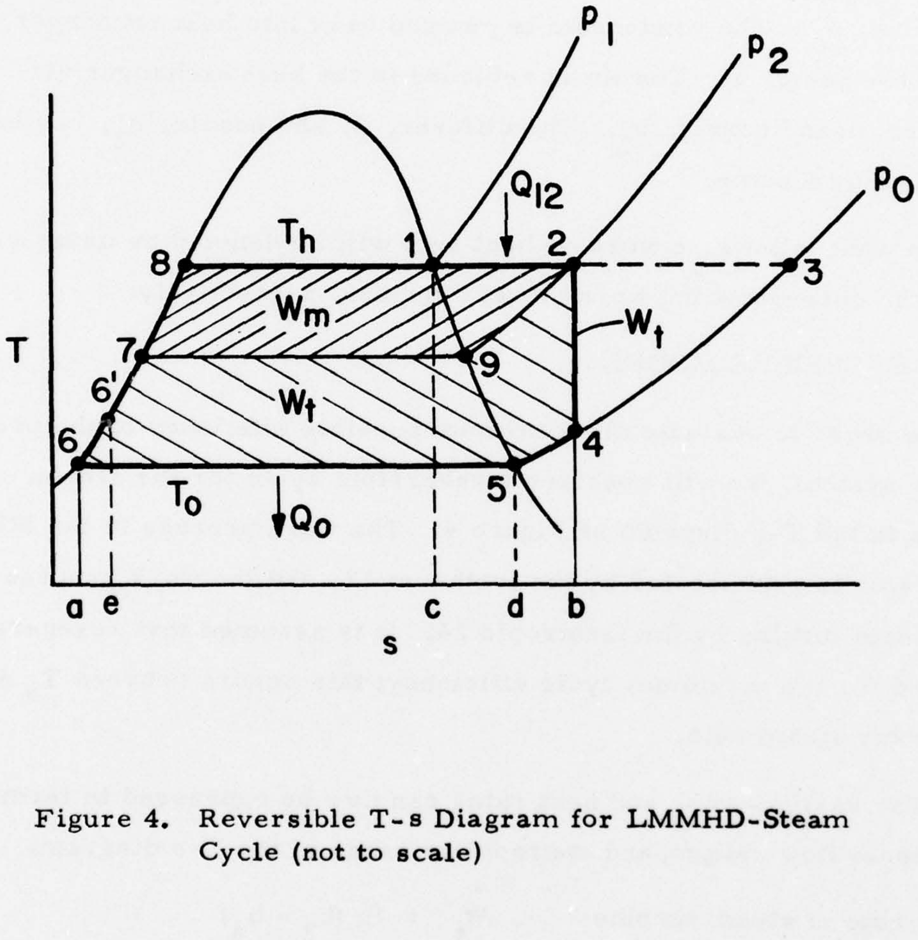

Figure 4. Reversible T-s Diagram for LMMHD-Steam Cycle (not to scale)

Work rate of MHD generator: $\dot{W}_m = \dot{m}\,[(h_1 - h_2) - T_h\,(s_1 - s_2)]$

$= \dot{m}$ (area 129781);

Total work rate: $\dot{W} = \dot{W}_t + \dot{W}_m$

$= \dot{m}[(h_1 - h_4) - T_h\,(s_1 - s_2)]$

$= \dot{m}$ (area 12456781);

Heat regeneration rate, vapor side: $\dot{Q}_r = \dot{m}\,(h_4 - h_5)$

$= \dot{m}$ (area 4bd54);

Heat rejection rate: $\dot{Q}_o = \dot{m}\,(h_5 - h_6)$

$= \dot{m}$ (area 5da65);

Pump power and heat regeneration rate, liquid side: $\dot{Q}'_r = \dot{m}\,(h_{6'} - h_6)$

$= \dot{m}$ (area 66'ea6) $= \dot{Q}_r$;

Primary heat addition rate: $\dot{Q}_h = \dot{m}\,(h_1 - h_{6'})$

$= \dot{m}$ (area 6'81ce6');

Total heat addition rate: $\dot{Q} = \dot{Q}_h + \dot{Q}_2$

$= \dot{m}\,[(h_1 - h_{6'}) + T_h\,(s_2 - s_1)]$

$= \dot{m}$ (area 6'8124be6');

Cycle efficiency: $\eta_{th} = \dfrac{\dot{W}}{\dot{Q}}$

$$= \frac{h_1 - h_4 + T_h\,(s_2 - s_1)}{h_1 - h_{6'} + T_h\,(s_2 - s_1)} .$$

The diagram shows that the choice of pressure p_2 (i.e., of state 2) determines the distribution of the total work, W, between the MHD generator (W_m) and the steam turbine (W_t). We shall describe this distribution by the work ratio:

$$r = \frac{\dot{W}m}{\dot{W}} = \frac{h_1 - h_2 + T_h (s_2 - s_1)}{h_1 - h_4 + T_h (s_2 - s_1)}$$

At one extreme, when state 2 coincides with the reference state 1 of the standard Rankine cycle, all the work is performed by the turbine ($r = 0$), and the T-s diagram is that of the reversible standard Rankine cycle (Figure 5). At the other extreme, when state 2 coincides with state 3 indicating that the steam expands in the MHD generator down to the condenser pressure p_o, all the work is done by the MHD generator ($r = 1$) and no steam turbine is required. The T-s diagram has the form shown in Figure 6.

Graphs of η_{th} vs. T_h for two condenser temperatures T_o = 125 and 193°F and $r = 0$, 0.7, and 1, are shown in Figure 7. The solid lines correspond to $r = 0$ (no MHD generator) whereas the dotted lines correspond to $r = 1$ (no steam turbine). As might be expected, the efficiency increases with T_h. For given T_h and T_o, the highest reversible efficiency is achieved if all the work is done by the MHD generator.

The gains in efficiency

$$\phi = \frac{\eta_{th}(r = 1)}{\eta_{th}(r = 0)} - 1,$$

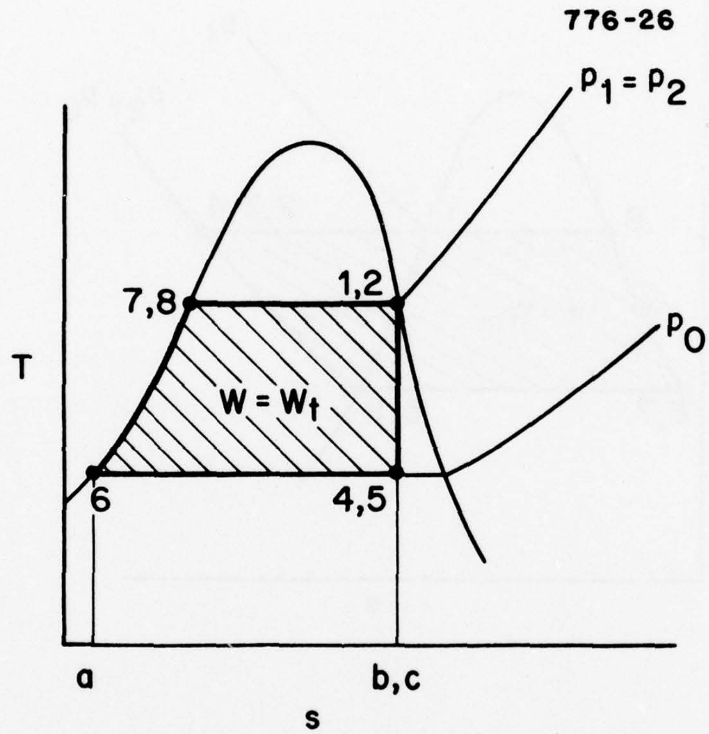

Figure 5. Reference Rankine Cycle (r = 0)

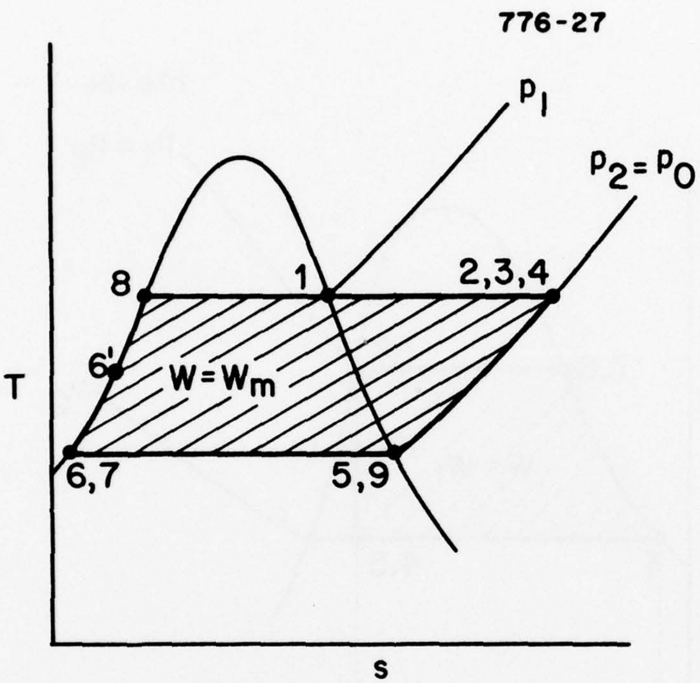

Figure 6. Reference MHD Generator Cycle (r= 0)

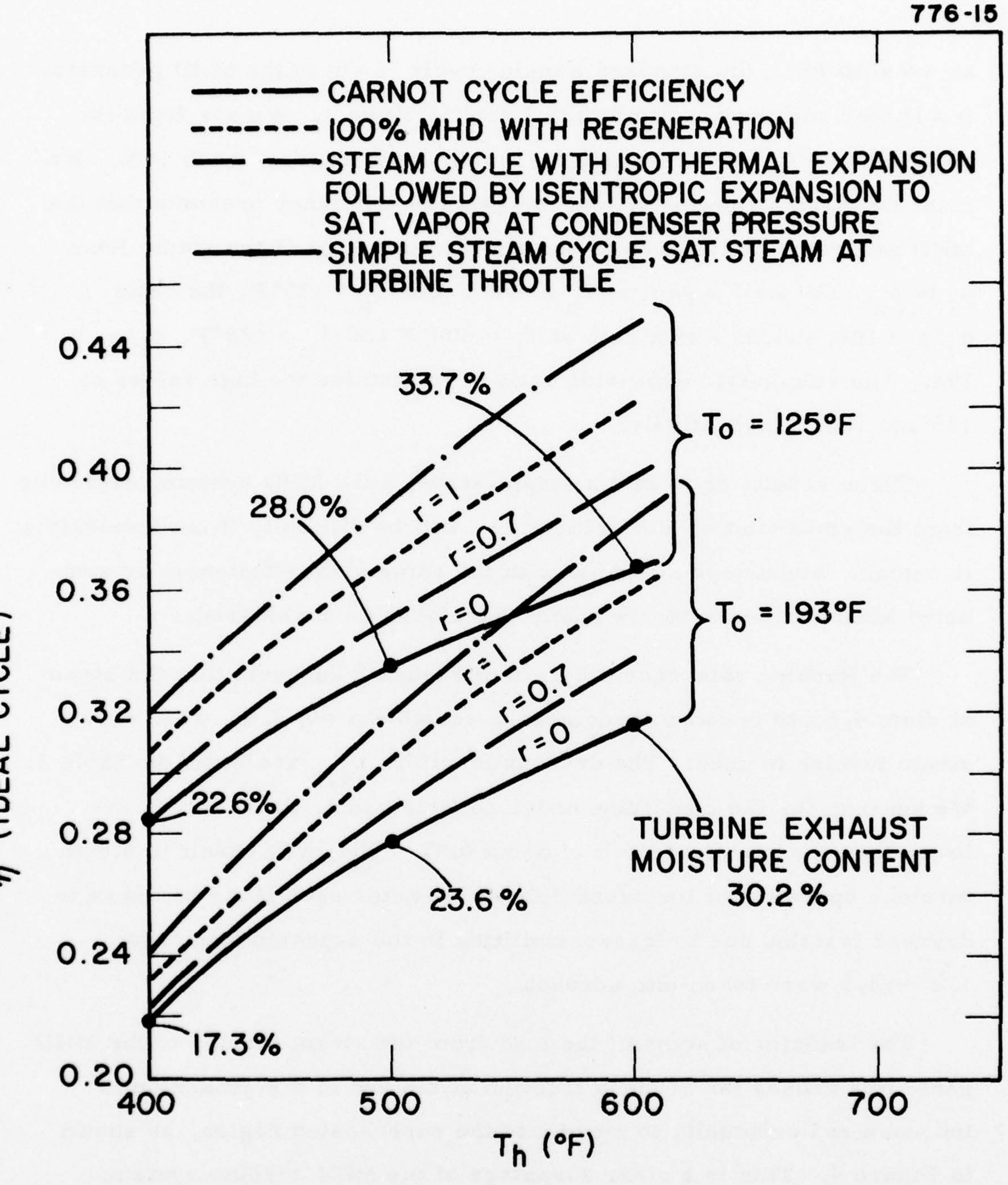

Figure 7. Ideal Cycle Efficiency Comparison

as we shift from the standard Rankine cycle ($r = 0$) to the MHD generator ($r = 1$), for various conditions are listed in Table 1. We see from the table that, at $T_h = 600°F$ and $T_o = 125°F$, the gain may reach 15%. We must emphasize, however, that the gains in efficiency presume that the MHD generator is designed to sustain the expansion of the steam from p_1 to p_o. For an 8% gain at $T_h = 400°F$ and $T_o = 125°F$, the ratio $p_1/p_o = 127$, and for a 15% gain at $T_h = 600°F$ and $T_o = 125°F$, $p_1/p_o = 794$. The volumetric expansion ratio v_3/v_1 attains the high values of 145 and 1251, respectively.

These results show that a single-stage, pure MHD system, desirable from the standpoint of ideal efficiency, will be difficult, if not impossible, to design. Multistage systems (with the same ideal efficiency) or combined MHD-turbine systems must, therefore, be considered.

The Rankine reference diagram of Figure 5 suggests that the steam at state 4, 5 can become unacceptably wet when $r = 0$; i.e., when only a steam turbine is used. The dryness fractions $x_{4,5}$ are listed in Table 2. We see that, for the conditions under consideration, these values are lower than the practical limit of about 0.87. They will result in steam turbines operating at low isentropic efficiencies even if the increase in dryness fraction due to irreversibilities in the expansion process $1,2 \rightarrow 4,5$ were taken into account.

The transfer of some of the load from the steam turbine to the MHD generator causes the dryness fraction at the end of the expansion to increase and eventually to move into the superheated region, as shown in Figure 4. This is a clear advantage of the MHD-turbine system. The dashed curves of Figure 7 correspond to a limiting case in which

TABLE 1

Fractional Gain, ϕ, in Reversible Efficiency
of MHD Cycle Over That of the
Corresponding Standard
Rankine Cycle

T_o \ T_h	400°F	500°F	600°F
125°F	0.074	0.107	0.150
193°F	0.058	0.092	0.139

TABLE 2

Dryness Fraction, $x_{4,5}$, After Ideal Expansion
From a Dry Saturated Condition
at State 1

	T_o \ T_h	400°F	500°F	600°F
$x_{4,5}$	125°F	0.774	0.720	0.663
	193°F	0.827	0.764	0.698

states 4 and 5 in Figure 4 are both on the saturated vapor boundary so that $x_{4,5} = 1$. Values of the efficiency gain, ϕ, and the work ratio, r, for this case are listed in Table 3.

As the load is transferred from the turbine to the MHD generator, and state 4 in Figure 4 moves into the superheated region, it becomes possible to apply heat regeneration as we have already indicated.

Finally, the dash-dot curves in Figure 7 represent the Carnot efficiency between T_h and T_o. We see that a reversible regenerative, pure MHD cycle comes quite close to the Carnot cycle.

The numerical results described in Figure 7 together with Tables 1 to 3 lead to the conclusion that the transfer of load from the turbine to the MHD generator is, generally speaking, advantageous. In the proposed scheme, the MHD generator replaces the high-pressure section of the turbine. Since the size of the turbine is governed by the volumetric flow rate, the corresponding reduction in turbine size will be less in proportion than the ratio of the two quantities of work.

Given that the steam turbine in a combined system operates at low pressures and large specific volumes, it appears useful to investigate whether it would be advantageous to replace the steam in this cycle by a refrigerant or a hydrocarbon as is done in the so-called bottoming cycles.

Though it provides a good starting point, the reversible model is not by itself the decisive factor in the evaluation of a power system because of differences in the effects that irreversibilities may have on different reversible cycles. To proceed with the evaluation, we must, therefore, consider irreversibilities.

TABLE 3

Fractional Gain, ϕ, in Reversible Efficiency and Work Ratio r for Combined MHD-Turbine Cycle with Expansion to Dry Saturated Condition ($x_{4,5} = 1$)

	T_o \ T_h	400°F	500°F	600°F
ϕ	125°F	0.029	0.051	0.083
	193°F	0.018	0.038	0.068
r	125°F	0.71	0.69	0.68
	193°F	0.67	0.65	0.63

C. IRREVERSIBLE MODEL

The results obtained from the reversible model indicate that a practical system is likely to consist of both an MHD generator and a turbine (not necessarily steam-driven). In order to evaluate the performance of such a system (efficiency, work ratio, and thermodynamic states), we must include in the analysis the tin circuit, irreversibilities, and the development and dissipation of kinetic energy in the two fluids. The inclusion of the tin circuit presents no difficulties, and the other effects can be considered by assuming reasonable efficiencies and pressure drops in the usual manner. To be sure, this requires that eventually we must be able to prove that components with the assumed performance can realistically be manufactured. Numerical values for efficiencies, etc., will be given in the next section. Here we merely list the factors that must be considered in order to arrive at an estimate of the effects of irreversibility.

- Efficiency of:
    - MHD generator
    - Water pump
    - Mixture nozzle n_1
    - Liquid metal nozzle n_2
    - Liquid metal diffuser
    - Steam turbine
- Pressure drops across:
    - Steam heater
    - Mixer
    - Liquid metal heater

- Liquid metal return duct
- Condenser

The inclusion of the irreversibilities will modify the conclusions derived from the analysis of the reversible model. In particular, it will be useful to investigate whether the overall efficiency, η, of the power system shows a maximum with respect to the work ratio, r, and whether this maximum is flat. Furthermore, it would be interesting to discover whether such an optimum design leads to wet, dry or superheated steam after the expansion in the turbine.

D. REVIEW OF ANL THERMODYNAMIC MODEL (CODE OFFDES)

For the review of the ANL thermodynamic model, we considered the computer results for a specific set of conditions of operation of the power system. We will call this the reference design; we give the input numerical values and the computer printout in the Appendix.

We performed an availability flow analysis of the reference design and compared our hand calculations with the computer results. From this comparison, we concluded the following:

- The steam properties used in the ANL code are satisfactory.
- The thermodynamic relations used in the code for calculating changes of liquid metal enthalpy and entropy are correct.
- The overall numerical results from the code are in good agreement with those of the hand calculations.

In addition, we have the following comments about specific relations in the code.

1. Void Fraction

The expression for the void fraction α can be derived as follows:

We will denote mass flow rate by $\dot{m}$ (lb/sec), density by ρ (lb/ft^3), velocity by u (ft/sec), flow area by A (ft^2), gas by subscript g, and liquid by subscript ℓ. Then the mass flow rates are

$$\dot{m}_g = \rho_g u_g A_g,$$

$$\dot{m}_\ell = \rho_\ell u_\ell A_\ell,$$

and the void fraction α is

$$\alpha = \frac{A_\ell}{A_g + A_\ell}$$

$$= \frac{\dot{m}_g/\rho_g u_g}{(\dot{m}_g/\rho_g u_g) + (\dot{m}_\ell/\rho_\ell u_\ell)}$$

$$= \frac{1}{1 + (\dot{m}_\ell \rho_\ell / \dot{m}_g \rho_g)\, k},$$

where $k = u_g/u_\ell$ = the slip ratio.

The expression for α is that used in the ANL code.

2. LMMHD Generator Analysis

The analysis of the LMMHD generator requires the following steps:

a. Calculate the isentropic work (or enthalpy change) of the tin-steam mixture through the generator, with specified

inlet pressure and temperature and specified exit pressure.

b. Calculate the actual work by multiplying the isentropic work by a specified generator efficiency.

c. Calculate the enthalpy of the tin-steam mixture at the exit of the generator by using the actual work.

d. Determine the exit temperature from the generator by using the exit pressure and enthalpy of the mixture and thermodynamic properties of tin and steam.

In these calculations, it is assumed that the fluid velocity through the generator is constant; i.e., that no change in kinetic energy occurs.

In all component calculations following, we use subscript "1" to denote the state of the working fluid entering the component, and subscript "2", the state of the working fluid exiting the component. Subscripts "ℓ" and "g" are used for tin and steam, respectively.

For the reference design, our hand calculation for the MHD generator gave results close to those of the ANL computer code. For steam properties, we used Keenan and Keyes. For changes in enthalpy and entropy of tin, we used the following expressions:

$$(h_2 - h_1)_\ell = \left[C_p (T_2 - T_1) + \overline{V} (1 - \beta_t T_2)(p_2 - p_1) \frac{144}{778}\right]_\ell;$$

$$(s_2 - s_1)_\ell = \left[C_p \, \mathrm{Th} \frac{T_2}{T_1} - \overline{V}\beta_t (p_2 - p_1) \frac{144}{778}\right]_\ell,$$

where h = enthalpy, Btu/lb

s = entropy, Btu/lb-R

C_p = tin heat capacity, assumed constant

$\beta_t = \frac{1}{\overline{V}}\left(\frac{\partial \overline{V}}{\partial T}\right)_p$, R^{-1}

$\overline{V}$ = specific volume of tin, ft^3/lb

T = temperature, R

p = pressure, psia.

The values of C_p and β_t used by ANL for molten tin are

C_p = 0.0615 Btu/lb-R

$\beta_t = 1.57 \times 10^{-5} R^{-1}$.

It is noteworthy that the value of the heat capacity of tin is somewhat questionable. For example, Barin and Knacke* give a value of 0.0577 Btu/lb-R for liquid tin instead of 0.0615 used by ANL. Before any extensive experimental program is begun, this discrepancy should be resolved. It can, however, be confidently stated that the discrepancy has no significant bearing on the analysis in this report.

The MHD generator power is calculated as

$$\dot{W}_m \left(\frac{Btu}{sec}\right) = \dot{m}_\ell (h_1 - h_2)_\ell + \dot{m}_g (h_1 - h_2)_g.$$

3. Liquid Metal/Steam Nozzle

For this component

$$\dot{m}_\ell (h_2 - h_1)_\ell + \dot{m}_g (h_2 - h_1)_g + \left(\frac{1}{2}\right)(\dot{m}_\ell)(U_2^2 - U_1^2)_\ell + \left(\frac{1}{2}\right)(\dot{m}_g)(U_2^2 - U_1^2)_g = 0,$$

*I. Barin and O. Knacke, Thermochemical Properties of Inorganic Substances, Springer-Verlag, Berlin/Heidelberg, 1973.

or,

$$\dot{H}_2 - \dot{H}_1 + \left(\frac{1}{2}\right)(\dot{m}_\ell)\,(U_2^2 - U_1^2)_\ell + \left(\frac{1}{2}\right)(\dot{m}_g)\,(U_2^2 - U_1^2)_g = 0,$$

where $\dot{H}$ denotes total enthalpy rate, tin plus steam. In other words, the total enthalpy change is converted into the kinetic energy of tin and steam. With given nozzle inlet pressure, p_1, and temperature, T_1, and nozzle exit pressure, p_2, the isentropic expansion is first calculated to give the isentropic rate of enthalpy change $\dot{H}_{2s} - \dot{H}_1$, where subscript "s" is used to denote the state reached at the end of the isentropic expansion. Then, the actual total enthalpy rate $\dot{H}_2 - \dot{H}_1$ is calculated by means of the relation

$$\dot{H}_2 - \dot{H}_1 = \eta_{noz}\,(\dot{H}_{2s} - \dot{H}_1),$$

where η_{noz} is the efficiency of the nozzle.

From the exit enthalpy and pressure, the exit temperature can then be determined from the steam and tin thermodynamic properties. Substituting the slip ratio, $k = u_g/u_\ell$, and the nozzle efficiency in the enthalpy balance we find the nozzle exit velocity:

$$U_{2\ell} = \sqrt{\frac{(\eta_{noz})\,(\dot{H}_{2s} - \dot{H}_1)\,(2)\,(32.2)\,(778)}{\dot{m}_\ell + k^2\,\dot{m}_g} + U_{1\ell}^{\,2}}\;.$$

This expression is identical to that in the ANL code, and a hand calculation agreed with the computer result where $U_{2\ell}$ = VEL-NØZ.

The ANL program incorporates a Mach number, M, which can be specified for the nozzle. The Mach number is calculated as follows:

$$M = \frac{U_{2\ell}}{C},$$

where

$$C = \sqrt{\frac{(p)\ (\dot{m}_\ell/\dot{m}_g)}{(\rho_\ell)\ (\alpha)\ (1-\alpha)\left(1+\frac{\dot{m}_\ell}{\dot{m}_g}\right)}}\ .$$

We have not investigated this calculation, since it does not affect the cycle efficiency for a specified input value of the nozzle efficiency. However, since presumably the reason for the calculation is to provide for realistic nozzle conditions in the cycle calculations, we would suggest that this effect be further evaluated and a realistic Mach number or other limit be incorporated in the program to ensure that unrealistic nozzle conditions are not used.

4. Liquid Metal/Steam Separator

For the separator, the pressure drop is taken as zero and the liquid kinetic energy recovery factor, η_{sep}, specified. This recovery is calculated for the tin as:

$$(h_2 - h_1)_\ell + \frac{1}{2}(U_2^2 - U_1^2)_\ell = 0;$$

$$\eta_{sep} = \frac{U_2^2}{U_1^2}\ ;$$

$h_2 - h_1 = C_p\ (T_2 - T_1)$ for constant pressure,

or

$$T_2 = T_1 + \frac{1}{2} U_1^2 \left(\frac{1 - \eta_{sep}}{C_p} \right) .$$

Basically, it is assumed that all of the lost kinetic energy goes to heating of the fluid.

For the gas or steam side, the gas exit temperature is set equal to the mixture inlet temperature. It is thus assumed that the gas kinetic energy is lost and not used to heat the gas. Although this effect can easily be calculated, the kinetic energy of the steam is very small relative to the enthalpy changes of the steam through the turbine. Therefore, neglecting the recovery of the steam kinetic energy in the separator is reasonable.

5. Liquid Metal Pump

The program includes provision for a liquid metal pump as either a replacement or a supplement for the liquid metal/steam nozzle and liquid metal diffuser. The pump work should be calculated as follows:

$$-\overline{W} = h_2 - h_1 \text{ Btu/lb} ;$$

$$\eta_p = \frac{h_{2s} - h_1}{h_2 - h_1} .$$

Thus,

$$-\overline{W}_p = \frac{1}{\eta_p} (h_{2s} - h_1).$$

Now,

$$dh = C_p dT + \overline{V}\,[1 - \beta_t T]\, dp\ ,$$

$$ds = C_p \frac{dT}{T} - \overline{V}\beta_t dp\ ,$$

where

$$\beta_t = \frac{1}{\overline{V}}\left(\frac{\partial \overline{V}}{\partial T}\right)_p .$$

For isentropic compression, ds = 0 and

$$C_p dT = \overline{V} T \beta_t dp\ .$$

Substituting this in the enthalpy relation we find

$$(dh)_s = \overline{V} T \beta_t dp + \overline{V} dp - \overline{V} T \beta_t dp$$

$$= \overline{V} dp.$$

Assuming constant $\overline{V}$, we find that the pump work is given by the relation

$$-\overline{W}_p = \frac{1}{\eta_p}\,\overline{V}\,[p_2 - p_1] = h_2 - h_1\ .$$

The temperature rise across the pump is given by

$$T_2 - T_1 = \frac{\overline{V}[p_2 - p_1]}{\eta_p\, C_p}\left[1 - \eta_p\,(1 - T\beta_t)\right] .$$

The pump work expression for the liquid metal pump in the ANL code is incorrect in that it includes the term $(1 - \beta_t T)$. Numerically this term makes only a small correction of less than 2% of the pump work; therefore, the code calculations are satisfactory.

6. Liquid Metal Diffuser

For the LM diffuser,

$$h_2 - h_1 + \frac{1}{2}(U_2^2 - U_1^2) = 0,$$

$$\eta_{diff} = \frac{h_{2s} - h_1}{h_2 - h_1} .$$

Thus,

$$\frac{1}{\eta_{diff}}(h_{2s} - h_1) = \frac{1}{2}(U_1^2 - U_2^2) .$$

For an isentropic diffuser,

$$dh = \overline{V} dP.$$

Thus,

$$\frac{\overline{V}}{\eta_{diff}}[p_2 - p_1] = \frac{1}{2}(U_1^2 - U_2^2),$$

or

$$p_2 = \frac{1}{2}\frac{(U_1^2 - U_2^2)(\eta_{diff})}{\overline{V}} + p_1 .$$

The temperature is obtained from the enthalpy relation as

$$T_2 = \frac{1}{2}(U_1^2 - U_2^2)\left(\frac{1}{C_p}\right) + \overline{V}\left[1 - \beta_t T_1\right]\left[p_1 - p_2\right]\left[\frac{1}{C_p}\right] .$$

These expressions are in agreement with those of the ANL code. In the code calculation, it is assumed that the liquid metal velocity leaving the diffuser is 1/10 of the velocity in the MHD generator.

7. Liquid Metal Heat Exchanger

For the LM heat exchanger,

$$(h_2 - h_1)_\ell = \left[C_p (T_2 - T_1) + \overline{V} (1 - \beta_t T_2) (p_2 - p_1)\right]_\ell .$$

The entering velocity is equal to the exiting velocity.

8. Liquid Metal Nozzle/Mixer

For this component, the ANL code assumes that the temperatures of both the tin and the steam are constant, with the steam having no pressure drop and the tin having a conversion of pressure head into kinetic energy. With this isothermal assumption, and neglecting the kinetic energy change of the steam, the enthalpy change of the steam is zero and the enthalpy change is due only to the tin. Thus, for the tin

$$(h_2 - h_1)_\ell + \frac{1}{2} (U_2^2 - U_1^2)_\ell = 0; \quad \eta_{noz} = \frac{h_2 - h_1}{h_{2s} - h_1} .$$

For an isentropic change

$$h_{2s} - h_1 = \overline{V} [p_2 - p_1];$$

therefore,

$$\eta_{noz} \overline{V} [p_2 - p_1] + \frac{1}{2} (U_2^2 - U_1^2) = 0,$$

or

$$p_2 - p_1 = \frac{1}{2} \frac{U_2^2 - U_1^2}{\eta_{noz} \overline{V}} .$$

This is equivalent to the equation in the ANL code with $U_1 = 0.1\,U_2$ or $U_2^2 - U_1^2 = U_2^2 - 0.01\,U_2^2 = 0.99\,U_2^2$ and the Δp's for the liquid metal heat exchanger, return pipe, and mixer subtracted from the LM diffuser exit pressure to give p_1.

Although the above relations are not strictly thermodynamically correct, they provide a close numerical approximation to the thermodynamically proper treatment and should be adequate for cycle efficiency evaluations.

9. Steam Calculations

The steam calculations are standard. Spot checking of steam properties and steam calculations in the ANL code disclosed no discrepancies or errors.

10. Summary

Though one could argue with the strict thermodynamic accuracy of certain parts of the ANL code, none of the deviations are important numerically. We believe that the computer model is more than adequate for reliable cycle efficiency calculations. Of much greater importance than the minor changes to the program which could be made (some of which are debatable) are the more serious questions of the validity of the assumptions made on the component efficiencies. It is on the component performance that future effort should be concentrated rather than more-or-less cosmetic changes to the ANL thermodynamic cycle computer calculation.

E. SENSITIVITY ANALYSIS

To provide an indication of the effect of various parameters on the cycle efficiency, ANL performed several cycle calculations using the

ANL computer program and parameter values specified by the evaluation team. The approach followed was to define a reference case representative of the system constraints. Then the value of each parameter was varied from that used in the reference case to provide an indication of the sensitivity of the cycle efficiency to that individual parameter. The parameters specified for the reference case are listed in Table 4. The split of the load between the LMMHD generator and the steam turbine, the cycle efficiency, and fraction of Carnot efficiency achieved in the reference case and the perturbed cases are listed in Table 5. From inspection of Table 5, we conclude the following:

- The condensing temperature is the most sensitive parameter affecting the cycle efficiency:

  $$\frac{\Delta \eta}{\Delta T_{cond}} = -8.68 \times 10^{-4}\ 1/°F.$$

  It should be noted that the effect of a condenser pressure drop is equivalent to a corresponding change in condensing temperature.

- The maximum cycle temperature (or mixer temperature) is the next most sensitive parameter:

  $$\frac{\Delta \eta}{\Delta T_{mix}} = 4.93 \times 10^{-4}\ 1/°F.$$

  These sensitivities correspond closely to values estimated from the approximate expression for Rankine-cycle efficiency.

  $$\eta \simeq 0.7 \eta_c,$$

TABLE 4

PARAMETERS FOR REFERENCE CASE

Temperature from mixer 425°F

Pressure from mixer (at steam saturation) = 325.92 psia*

Inlet void fraction to LMMHD generator = 0.650*

Exit void fraction from LMMHD generator = 0.85

Slip ratio = 1.00

Steam turbine efficiency = 0.85

LMMHD generator efficiency = 0.80

LMMHD stages = 1.00

Separator recovery ratio = 0.95

Δp of LM heat exchanger = 25 psi

Δp of mixer = 5 psi

Δp of condenser = 0 psi

Number of feedwater heaters = 0

$\eta_{diff} = 0.8$

$\eta_{noz} = 0.9$

$\eta_{water\ pump} = 0.72$

Condensing temperature = 125°F (1.94 psia)

*Selected by ANL as appropriate for reference case.

TABLE 5

SENSITIVITY OF CYCLE EFFICIENCY TO PARAMETER VARIATION

Parameter Varied	Change in Parameter	Power Output,** Btu/sec: LMMHD Gen. Output	Power Output,** Btu/sec: Steam Turbine Output	Mass Ratio lb/sec Tin Per lb/sec Steam	Net Cycle Efficiency	Fraction of Carnot Efficiency	Comments
Reference Case	-	337	667	336	0.244	0.719	See Table 4 for Component Efficiency Values and Appendix for Computer Printout
Temperature from Mixer (Maximum Cycle Temp.)	From 425° F to 500° F	251	757	159	0.281	0.718	$\Delta\eta/\Delta T_h = -4.93 \times 10^{-4}\ \frac{1}{°F}$
Condensing Temperature	From 125° F (1.94 psia) to 193° F (9.96 psia)	471	536	336	0.185	0.704	$\Delta\eta/\Delta T_c = -8.68 \times 10^{-4}\ \frac{1}{°F}$
Cycle	Steam System Only with No LMMHD	0	1005	0	0.254	0.749	Turbine throttle was saturated steam at 425° F (325.92 psia); this resulted in a moisture content in the turbine exhaust steam of 19%, which is greater than acceptable. Reducing the cycle pressure to lower this moisture content to the maximum acceptable level of 13% will reduce the efficiency of the steam-only system appreciably; see Section II.G.
	Single Stage LMMHD w/o Steam Turbine	1006	0	762	0.179	0.530	Gen. Void Fraction; 0.45 → 0.88; Exit P from Generator = 41.0 psia Condenser P = 2.06 psia (127° F) Regenerator Vapor Side ΔP = 5 psi
	Two-Stage LMMHD w/o Steam Turbine***	1007	0	268	0.170	0.523	Gen. Void Fraction; 0.700 → 0.845 and 0.700 → 0.914 Gen. Pressures; 325.92 → 150.0 psia and 89.77 → 20.0 psia Condenser P = 2.783 psia (138° F) Regenerator ΔP = 5 psi***
Slip Ratio	From 1.0 to 1.1	329	676	305	0.245	0.723	
ΔP of Mixer	From 5 to 10 psi	338	667	336	0.243	0.718	
	From 5 to 0 psi	336	668	336	0.245	0.721	
ΔP of LM Heat Exchanger	From 25 to 0 psi	334	671	336	0.247	0.727	
ΔP of Condenser	From 0 to 1.0 psi	337	667	336	0.239	0.650	
Diffuser Efficiency	From 0.8 to 0.6	357	647	336	0.230	0.677	
LM Circulation	LM Pump with 0.8 Efficiency Instead of Nozzle/Diffuser	350 Gross (Less 153 for LM Pump)	807	336	0.245	0.721	
LMMHD Generator Efficiency	From 0.8 to 0.7	308	697	336	0.236	0.696	
Inlet Void Fraction of Generator	From 0.65 to: 0.75 0.65 0.55 0.45 0.35	169 332 473 622 904	835 672 532 382 100	208 336 510 763 1160	0.249 0.244 0.238 0.228 0.206	0.733 0.719 0.700 0.672 0.607	Exit Void Fraction = 0.83 → 0.85 for all cases. Calculations with LM Pump instead of Nozzle/Diffuser gave almost identical cycle efficiencies
Exit Void Fraction of Generator	From 0.85 to: 0.95 0.76	590 168	414 837	336 336	0.246 0.244	0.725 0.718	Inlet Void Fraction = 0.65 as for Reference Case
Separator Recovery Ratio	From 0.95 to: 0.80 0.50	348 402	657 604	336 336	0.236 0.204	0.696 0.602	The cycle efficiency is less sensitive to the separator recovery factor if a LM pump is used, the corresponding efficiencies being 0.243 (for 0.80 factor) and 0.239 (for 0.50 factor).
Use of 2 LMMHD Stages	From 1 to 2 LMMHD Stages	636	369	267	0.224	0.661	Generator Void Fraction; 0.700 → 0.845 and 0.700 → 0.914 Generator Pressure; 325.92 → 150.0 psia and 89.77 → 20.0 psia Condenser P = 1.942 psia (125° F)
		329	676	208	0.235	0.693	Generator Void Fraction; 0.75 → 0.80 and 0.75 → 0.80 Generator Pressure; 325.92 → 185.0 psia and 130.5 → 80.0 psia Condenser P = 1.942 psia (125° F)

*Based on computer calculations performed at Argonne National Laboratory by Dr. E. Pierson specifically for this evaluation.

**Total Net Power Output = 1000 Btu/sec for all cases.

***Reducing the regenerator vapor-side ΔP from 5 psi to 0 psi results in a substantial increase in the cycle efficiency to ~0.21, though with a high exit void fraction (~0.95) from the second LMMHD stage.

where η_c is the Carnot cycle efficiency given by

$$\eta_c = \frac{T_h - T_o}{T_h}$$

and where T_h and T_o are the heat source and heat sink temperatures, respectively.

$$\eta \simeq 0.7 \frac{T_h - T_o}{T_h};$$

$$\frac{\partial \eta}{\partial T_o} \simeq -0.7 \frac{1}{T_h} = -\frac{0.7}{885°R} = -7.9 \times 10^{-4}\ 1/R;$$

$$\frac{\partial \eta}{\partial T_h} \simeq +0.7 \frac{T_o}{T_h^2} = 0.7 \frac{585°R}{(885°R)^2} = +5.2 \times 10^{-4}\ 1/R;$$

- With respect to the LMMHD parameters, reducing the diffuser efficiency from 0.8 to 0.6 (25% reduction) reduces the cycle efficiency from 0.244 to 0.230 (6% reduction). This relatively large effect is due to the large pumping power required for the liquid metal circulation. Thus, for the case with a liquid metal pump with 0.8 efficiency, the LMMHD generator work output is 350 Btu/sec with 153 Btu/sec of this output required for circulation of the liquid metal. It should be noted that, if a sufficiently high diffuser efficiency cannot be achieved, this problem can be circumvented by the use of a liquid metal pump in place of the nozzle/diffuser.
- The slip ratio, mixer Δp, and LM heat exchanger Δp have a small effect on the cycle efficiency over the range covered.

- The steam only system efficiency is basically the same as that of the LMMHD/steam turbine system (see Table 6 also). Thus, attaining high efficiency must not be considered the main purpose for development of the low-temperature LMMHD system, since as high an efficiency can be achieved with a conventional steam system.

- The efficiency of the LMMHD system without the steam turbine is lower than that of the system with a steam turbine. The basic limitations on efficiency of the pure LMMHD system (steam as working fluid) is the maximum exit generator void fraction at which the generator can operate with high efficiency. If there were no restrictions on this parameter, the cycle efficiency of the LMMHD could be as high as either the steam only cycle or the LMMHD/steam turbine combined system. As suggested earlier, a working fluid other than steam might alleviate this limitation on the LMMHD only system.

F. THERMODYNAMIC AVAILABILITY FLOW

The availability flow through the reference design system is shown in Figure 8. Each availability loss is due to the irreversibility of the corresponding component.

The availability flow is divided into the two circuits, steam and tin. For convenience, the availability of the water into the heat exchanger is taken equal to zero. The change in availability between states 1 and 2 of a fluid per unit flow rate is defined as

TABLE 6

COMPARISON OF STEAM CYCLE AND LMMHD-STEAM CYCLE EFFICIENCIES
Steam Cycle Conditions

A-893

Maximum Cycle Temperature °F	Condensing Temperature °F	System	Cycle Efficiency %	Percent of Carnot	Turbine Exhaust Moisture Content %
425	125	Simple steam	22.0	0.647	13.0
		Steam with one feed-water heater	23.2	0.683	13.0
		Steam with one stage reheat	25.0	0.736	9.0
		Steam with one stage reheat and one feed-water heater	26.0	0.765	9.0
		LMMHD-Steam ANL calculation (Reference case)	24.4	0.718	7.5
500	193	Simple steam	21.2	0.663	13.0
		Steam with one feed-water heater	22.4	0.700	13.0
		Steam with one stage reheat	23.2	0.725	11.0
		Steam with one stage reheat and one feed-water heater	24.2	0.757	11.0
		LMMHD-Steam ANL calculation	22.9	0.716	9.5

777-7

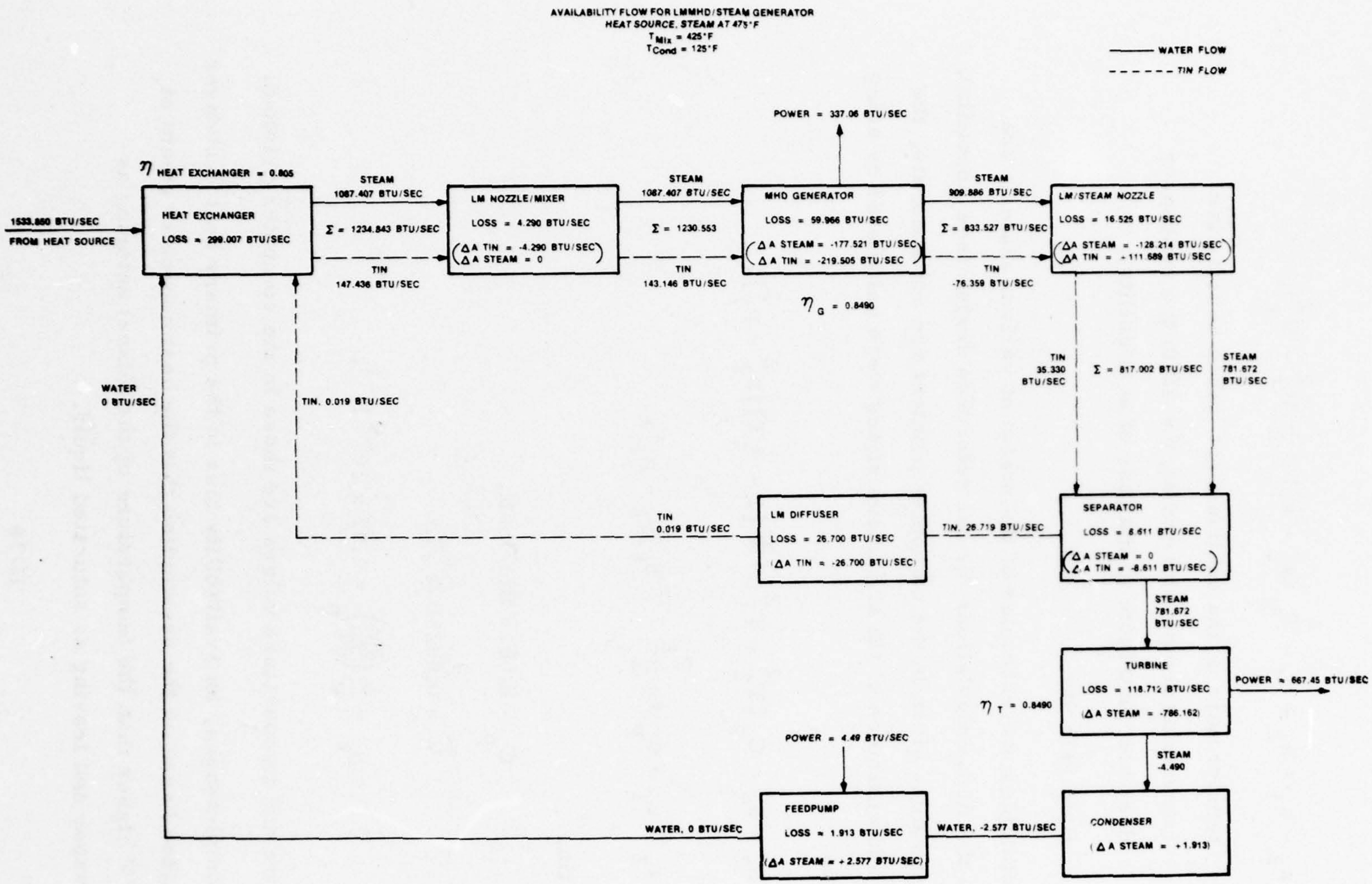

Figure 8. Availability Flow for LMMHD/Steam Generation

$$a_2 - a_1 = h_2 - h_1 - T_o (s_2 - s_1) + \frac{1}{2}(U_2^2 - U_1^2),$$

where T_o = temperature of the environment (taken here as the condensing temperature; thus, the ΔT in the condenser does not contribute to the loss of availability);

U = fluid velocity.

The enthalpies and entropies of the steam at various states were taken from the ANL calculations for the reference design (see Appendix). Because the values given in the computer printout are not accurate, the enthalpies and entropies of tin at various states were calculated by using the relations

$$h_2 - h_1 = C_p [T_2^2 - T_1^2] + \overline{V} [1 - \beta_t T] [p_2^2 - p_1^2],$$

$$s_2 - s_1 = C_p \ln \frac{T_2}{T_1} - \overline{V} \beta_t [p_2 - p_1],$$

where, for tin,

$$C_p = 0.0615 \text{ Btu/lb-R},$$

$$\overline{V} = 0.00235 \text{ ft}^3\text{/lb},$$

$$\beta_t = \frac{1}{\overline{V}} \left(\frac{\partial \overline{V}}{\partial T}\right)_p = 1.57 \times 10^{-5} \text{ R}^{-1},$$

and pressure and temperature values are those in the computer printout.

For completeness, an availability loss in the primary heat exchanger was calculated based on the assumption that the heat source is steam at 475°F (50°F higher than the temperature of the mixer) entering as saturated vapor and leaving as saturated liquid.

We see from Figure 8 that:

a. The availability losses in the LM nozzle/mixer, LM/steam nozzle separator, and LM diffuser are relatively small, amounting to a total of 56.13 Btu/sec or only 4.5% of the availability input to the steam and tin circuits. The major losses are in the LMMHD expander (60.0 Btu/sec) and in the steam turbine (118.7 Btu/sec).

b. The importance of the heat exchanger ΔT is illustrated by the large loss of availability for this component (299 Btu/sec for 475°F heat source temperature). The condenser ΔT would have a similar large effect.

c. The availability efficiency of the steam turbine is

$$\eta = \frac{\text{Power out}}{\text{Availability change}} = \frac{667.45}{786.16} = 0.85.$$

d. The availability efficiency of the LMMHD generator is

$$\eta = \frac{337.06}{397.02} = 0.85.$$

It is noteworthy that efficiency calculated on the basis of availability is numerically different than the customary efficiency, which is based on enthalpic considerations. Specifically, for a turbine:

$$\eta_{\text{availability}} = \frac{\Delta h_{\text{actual}}}{\Delta h_{\text{actual}} - T_o \, \Delta S_{\text{actual}}},$$

$$\eta_{\text{enthalpy}} = \frac{\Delta h_{\text{actual}}}{\Delta h_{\text{isentropic}}}.$$

G. COMPARISON OF LMMHD-STEAM AND ALL STEAM SYSTEMS

The efficiencies of several modified steam Rankine-cycle systems have been calculated for comparison with the efficiency of the reference design of the LMMHD-steam power system. The results are listed in Table 6.

The steam cycle calculations were made with a steam turbine efficiency of 85%, feedpump efficiency of 75%, no component pressure drops, and a maximum turbine exhaust moisture content of 13%. The LMMHD-steam reference design calculations include the pressure drops given in Table 4. The efficiency penalty associated with the pressure drops, however, is at most a few tenths of a percent. The 85% steam turbine efficiency has been assumed for consistency with the ANL calculations.

It is clear from the results given in Table 6 that the efficiency of the steam cycles is comparable to that of the LMMHD-steam system.

III. PROPOSED 2-STAGE, 1-MW DEMONSTRATION UNIT

This section of the report discusses the proposed 1-MW demonstration unit described in Report ANL/ENG-76-06 entitled "Tin-Steam MHD Power System," Volume II by M. Petrick, B.K. Snyder and L. Carlson, published in September 1975, and in particular on pp. 1-9 of the above ANL report. The discussion addresses itself to the conceptual design and does not attempt to analyze the proposed components in any detail.

The ANL proposal has two alternatives. Alternative 1 proposes to design and build a 2-stage pure MHD system operating between 1000 lbf/in^2 (T_h = 544.6°F) and 75 lbf/in^2 (T_o = 307.6°F), with an interstage pressure of 300 lbf/in^2 (417.3°F at saturation) and heat regeneration. Each stage is designed to produce 75 kW at 3 V and 25,000 A and to use cryogenic, superconducting magnets. The rating of 1 MW is based on the heat to be supplied to the electric boiler, which simulates a nuclear reactor. Alternative 2 proposes to construct the 1000 - 300 lbf/in^2 stage only.

The cycle analysis and discussion given in Section II provide a background against which these proposals can be understood.

The panel considers that the paramount task of a demonstration unit is to develop the new components; namely,

- the MHD generator
- the mixer and
- the separator.

The panel further considers that a quest for high overall efficiency, as opposed to high component efficiency, is premature. If new components of high efficiency can be developed, the whole scheme merits further attention even though ultimate success cannot be guaranteed. Failure to achieve success with only one element among those needed in the unit will doom the whole idea to failure. For these reasons, it appears that the concept itself should be changed from proposing to build a demonstration unit to proposing to construct a test facility for the new components.

If it is decided to construct a test facility, it would seem logical to disregard considerations of system efficiency completely in favor of making the facility as simple and as flexible as possible. The existing computer program for the system is adequate to monitor performance in the light of the progress achieved in the sequence of testing and improvement.

IV. CONCLUSIONS AND RECOMMENDATIONS

1. The efficiencies of the LMMHD/steam system and modified Rankine all steam systems are about equal (see Tables 5 and 6). It follows that the advantage of the LMMHD/steam system must be judged from features such as volume, weight, reliability, amount and size of rotating machinery, and noise level.

2. The efficiency of the pure LMMHD system (i.e., without the steam turbine) with steam as the thermodynamic working fluid is significantly lower than a pure steam system (see Table 5). Since an efficient, pure LMMHD system would be preferable by far to a combined LMMHD steam system with the same efficiency, a strong effort with this goal is recommended. Two approaches can be investigated:

   a. Use of many LMMHD generator stages for high-pressure ratio operation as required for steam as the thermodynamic working fluid (How many LMMHD generator stages can effectively be used with steam?).

   b. Search for an alternate thermodynamic working fluid to maximize the efficiency of the pure LMMHD system for 400-500°F heat source temperatures.

3. A 1-MW prototype system should not be constructed at this time.

4. We recommend that the program be continued as a basic technology development effort concentrating on component development with parallel systems analysis and optimization. The program should concentrate on the pure LMMHD system without a steam turbine. Areas which should be part of the technology program are:

a. LMMHD generator development and performance measurements - in particular, influence of variables such as velocity, inlet/outlet void fractions, gas pressure, bubble size, and degree of dispersion of gas in liquid.

b. Mixer development - How much power is required and what is the best technique to obtain optimum gas dispersion in liquid?

c. Separator development - The separator could represent an extremely important component for the success of the overall system. The carryover of the liquid metal in the form of droplets in the steam must be quite small, since the liquid metal droplets will freeze during either turbine expansion or regeneration/condensation.

d. Compatibility - The chemical compatibility of the liquid metal with the steam or other working fluid must be considered much more completely.

e. As suggested in recommendation 2.b, alternate thermodynamic working fluids, which could have better characteristics than steam for coupling to LMMHD, should be investigated with the primary goals of improving the efficiency of the pure LMMHD system and of reducing its size.

f. The question of part-load operation of the LMMHD system should be considered in terms of the method of operation and effect on efficiency.

5. While the technology development program is progressing, it is necessary to keep the final applications in mind in order to provide perspective to the technology development. Therefore, analyses for specific applications as a guide in specifying the technology development areas and approach should be expanded.

APPENDIX

REFERENCE CASE COMPUTER PRINTOUT

```
****** INPUT DATA ******

                                                 RUN ON    05-25-77   AT   10:20   HOURS
CYCLE_TYPE='RANKINE',DESIGN_POWER=1000.,
WORKING_FLUID='STEAM',
STEAM_TURB='YES',P_TURB=1.9420,

FEED_WATER='NO',
NUMBER_OF_FEEDWATER_HEATERS=2,

NUMBER_OF_GENERATORS=1,
P_MIX(1)=325.92,T_MIX(1)=885.,ALPHA_MIX(1)=0.65,
                T_MIX(2)=885.,ALPHA_MIX(2)=0.75,
P_GEN(1)=110.0,P_GEN(2)=10.,
GEN_EFF(1)=.80,GEN_EFF(2)=.80,
GEN_LIQ_VEL(1)=50.,GEN_LIQ_VEL(2)=50.,
K_MIX(1)=1.0,K_MIX(2)=1.0,K_GEN(1)=1.0,K_GEN(2)=1.0,

PUMP_OPTION='NOZZLE',MACH_LIMIT=5.0,
ETA_WPUMP=0.72,ETA_LMPUMP=0.80,ETA_DIFF=0.8,ETA_NOZ=0.9,
ETA_TURB=0.85,SEP_REC=.95,ETA_LMNOZ=.95,
DELTA_P_STMHX=0.0,DELTA_P_FEEDWTR_HTR=5.0,
DELTA_P_MIX=5.,DELTA_P_LMHX=25.,DELTA_P_RTNP=0.0,VEL_RTNP(1)=5.0,VEL_RTNP(2)=5.,
DELTA_P_COND=0.0,DELTA_P_STMREGEN=5.0,STMREGEN_EFFECTIVENESS=0.90,
PRINTED_OUTPUT='YES', NFLAG4=-1;

*** END OF INPUT DATA ***
```

```
    NUMBER_OF_FEEDWATER_HEATERS =     0

TIME_IS='102034111'      DATE_IS='770525';

N_POWER_CALC=              1;
MASSG(1)= 1.00000E+00    MASSG(2)= 1.00000E+02    MASSG(3)= 1.00000E+02    MASSG(4)= 1.00000E+02    MASSG(5)= 1.00000E+02;

T_MIX= 8.85000E+02       P_MIX= 3.25919E+02       ALPHA_MIX= 6.49999E-01   GEN_EFF(1)= 7.99999E-01  T_GEN= 8.79885E+02
P_GEN= 1.10000E+02       ALPHA_GEN= 8.57041E-01   HG_GEN= 1.23668E+03      SG_GEN= 1.65219E+00      GEN_POWER= 1.02940E+02
T_ISENTROPIC= 8.78670E+02                         MASSG= 1.00000E+00       MASSL= 3.35901E+02       X= 2.96822E-03;
T_NOZ= 8.76942E+02       P_NOZ= 5.71093E+01       VEL_NOZ= 1.08469E+02     TL_SEP(1)= 8.77133E+02   TG_SEP(1)= 8.76942E+02
P_SEP(1)= 5.71093E+01    SEP_LIQ_VEL(1)= 1.05723E+02                       MASSL= 3.35901E+02;
ENERGY BALANCE IS NOT CHECKED ON THE FIRST ITERATION

A SUMMARY OF THE ENERGY BALANCE FOLLOWS.
    COMPONENT          DELTA_E          FLUID
    GENERATOR 1 -1.0294516E+02  1.0294061E+02
    SEPERATOR 1 -1.8670654E-01  0.0000000E+00
    LM PUMP   1  0.0000000E+00  0.0000000E+00
    TURBINE     -2.0385181E+02  2.0385161E+02
    CONDENSER   -9.4561377E+02  9.4561377E+02
    WATER PUMP   1.3720551E+00 -1.3722095E+00
    PRIM HX      1.2518824E+03  1.2510337E+03

N_POWER_CALC=              2;
MASSG(1)= 3.27418E+00    MASSG(2)= 1.00000E+02    MASSG(3)= 1.00000E+02    MASSG(4)= 1.00000E+02    MASSG(5)= 1.00000E+02;

T_MIX= 8.85000E+02       P_MIX= 3.25919E+02       ALPHA_MIX= 6.49999E-01   GEN_EFF(1)= 7.99999E-01  T_GEN= 8.79885E+02
P_GEN= 1.10000E+02       ALPHA_GEN= 8.57041E-01   HG_GEN= 1.23668E+03      SG_GEN= 1.65219E+00      GEN_POWER= 3.37056E+02
T_ISENTROPIC= 8.78670E+02                         MASSG= 3.27418E+00       MASSL= 1.09980E+03       X= 2.96822E-03;
T_NOZ= 8.76942E+02       P_NOZ= 5.71093E+01       VEL_NOZ= 1.08469E+02     TL_SEP(1)= 8.77133E+02   TG_SEP(1)= 8.76942E+02
P_SEP(1)= 5.71093E+01    SEP_LIQ_VEL(1)= 1.05723E+02                       MASSL= 1.09980E+03;

HEAT_ACROSS_BOUNDARIES= 4.096117187500000E+03    HEAT_ADDED_TO_FLUID= 4.098886718750000E+03    NORM_EN_DIFF=-6.766125694493450E-04
EN_CONV= 9.99999E-03;

A SUMMARY OF THE ENERGY BALANCE FOLLOWS.
    COMPONENT          DELTA_E          FLUID
    GENERATOR 1 -3.3706079E+02  3.3705615E+02
    SEPERATOR 1 -6.1125183E-01  0.0000000E+00
    LM PUMP   1  0.0000000E+00  0.0000000E+00
    TURBINE     -6.6744678E+02  6.6744653E+02
    CONDENSER   -3.0961108E+03  3.0961108E+03
    WATER PUMP   4.4923553E+00 -4.4928637E+00
    PRIM HX      4.0988887E+03  4.0961172E+03
```

SUMMARY OF CALCULATIONS FOR RANKINE CYCLE USING MHD GENERATORS.

NUMBER OF GENERATOR STAGES =	1	
MIXER TEMPERATURE =	885.00	DEG R
MIXER PRESSURE =	325.92	PSIA
GAS FLOW RATE =	3.27	LB/SEC
WORKING FLUID=	STEAM	
FEED WATER HEATERS=	NO	
STEAM TURBINE=	YES	
DIFFUSER EFFICIENCY=	0.800	
NOZZLE EFFICIENCY=	0.900	
TURBINE EFFICIENCY=	0.850	
LM PUMP EFFICIENCY=	0.800	
WATER PUMP EFFICIENCY=	0.720	
STMREGEN EFFECTIVENESS=	0.900	
MIXER PRESSURE LOSS=	5.00	PSI
LM HEAT EXCH PRESSURE LOSS=	25.00	PSI
CONDENSER PRESSURE LOSS=	0.00	PSI
STEAM REGEN HX PRESSURE LOSS=	5.00	PSI
NET POWER OUTPUT =	1000.01	BTU/SEC
MHD POWER =	337.06	BTU/SEC
TURBINE POWER =	667.45	BTU/SEC
STMREGEN HOT SIDE ENTHALPY DROP=	0.00	BTU/SEC
CONDENSER HEAT REJECTED=	3096.11	BTU/SEC
LM CIRCULATING POWER =	0.00	BTU/SEC
GAS CIRCULATING POWER =	-4.49	BTU/SEC
THERMAL INPUT POWER =	4098.887	BTU/SEC
CARNOT EFFICIENCY =	0.339	
CARNOT EFFICIENCY RATIO =	0.719	
NET THERMAL EFFICIENCY =	0.2440	

SUMMARY OF NOZZLE STAGES.

STAGE NUMBER 1

MACH NUMBER AT NOZZLE EXIT =	1.1897	
NOZZLE EXIT AREA =	0.2941241	SQ.FT
NOZZLE EXIT VOID FRACTION =	0.92140	
MACH_LIMIT =	5.00	
EXTRA_LMPUMP =	NO	

SUMMARY OF GENERATOR STAGES.

STAGE NUMBER 1

GAS FLOW RATE =	3.27	LB/SEC
LIQUID FLOW RATE =	1099.80	LB/SEC
MHD POWER =	337.06	BTU/SEC
LM CIRCULATION POWER =	0.00	BTU/SEC
GENERATOR EFFICIENCY =	0.800	
GENERATOR ENTRANCE VOID FRACTION=	0.650	
GENERATOR EXIT VOID FRACTION=	0.857	
EXIT SLIP RATIO =	1.000	
INLET SLIP RATIO=	1.000	

SUMMARY OF THERMODYNAMIC STATE POINTS.

ELEMENT EXIT COND.	TEMPERATURE DEG R	PRESSURE PSIA	GAS ENTROPY BTU/LB-DEG R	GAS ENTHALPY BTU/LB	LIQUID ENTROPY BTU/LB-DEG R	LIQUID ENTHALPY BTU/LB	LIQUID VELOCITY FT/S
STAGE NUMBER 1							
MIXER	885.000	325.920	1.503	1203.765	0.417	54.563	
GENERATOR	879.885	110.000	1.652	1236.681	0.417	54.159	50.000
SEP.-LIQUID	877.134	57.109			0.417	53.967	105.723
NOZZLE	876.943	57.109	1.729	1242.551	0.417	53.956	108.470
LMDIFFUSER	877.896	479.037			0.417	54.190	5.000
LM HEAT EXCH.	885.000	449.037			0.417	54.614	5.000
SEPERATOR-GAS	876.943	57.109	1.729	1242.551			
TURBINE	584.795	1.942	1.791	1038.700			
CONDENSER	584.795	1.942	0.173	93.086			
WATER PUMP	585.333	330.920	0.174	94.458			
GASLOOP	585.333	330.920	0.174	94.458			

Addendum to the Report

"Thermodynamic Analysis of a Tin-Steam LMMHS System", by
Thermo Electron Corporation, July 29, 1977

Comments on two reports: (1) "The Nuclear Submarine Application of LM-MHD by J.L. Maskasky, Shearwater, Inc.; and (2) "An Analysis of the Nuclear Submarine Application of LM-MHD," by ANL, April 29, 1977.

The evaluation panel reviewed the report by Shearwater on the "Nuclear Submarine Application of LM-MHD," and the ANL response to this report.

The principal conclusions of the Shearwater report are:

1. That LM-MHD cycles may be potentially thermodynamically competitive with steam systems over the same temperature range.

Obviously, ANL agrees with this conclusion. The panel also reached the same conclusion.

2. That a pure LM-MHD cycle which fully utilizes the same temperature range as the reference system would require a tin inventory sufficiently large to reject the concept for submarines on that basis alone.

ANL disagrees with this conclusion. It performs a simple weight calculation that results in a reduction of 330 klb with respect to the Shearwater estimate. ANL expects further reductions with more detailed and optimized calculations.

In view of the present stage of development of various key components of the LM-MHD system, the panel feels that it is premature to judge definitively the potential applications of the system on the basis of either the Shearwater or the ANL weight estimates.

3. That the combined cycle which does not replace the steam turbine as a power converter, provides an efficiency gain insufficient to offset its forecast weight growth. As a result, the installation of the best combined cycle investigated would result in a larger, slower, and more costly submarine.

ANL's response to this criticism is that the system has not been subjected to design optimization.

The panel believes that a system optimization cannot even be initiated until better experimental information is accmulated about the key LM-MHD components, and the heat source conditions and the application of the system are better prescribed.

4. That there are serious containment and safety issues associated with the liquid tin, etc.

ANL questions this conclusion.

The panel does not have any expertise in this area, but finds ANL's arguments reasonable.